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COOLING TESTS OF AN AIR-COOLED ENGINE CYLINDER
WITH COPPER FINS ON THE BARREL

By J. C. Sanders, H. D. Wilsted, and
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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

ADVANCE CONFIDENTIAL REPORT

COOLING TESTS OF AN AIR-COOLED ENGINE CYLINDER
WITH COPPER FINES ON THE BARREL

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SUMMARY

Comparative cooling tests were run on two Wright C9GC (G-200) cylinders, one with the original steel fins and one with 1-inch spiral copper fins brazed on the barrel. The copper fins improved the over-all heat-transfer coefficient from the barrel to the air 115 percent. At take-off power the temperature of the barrel for the same cooling-air pressure drop decreased from 297° F to 213° F. No structural defect had developed after 60 hours of operation at brake mean effective pressures of 45 to 258 pounds per square inch.

The improvements determined by test were in reasonable agreement with the improvement computed from fin dimensions. Computations were therefore made to determine the improvement in cooling that could be obtained with copper and aluminum fins having the same weights as the original steel fins. In the ranges of practical fin sizes the copper and aluminum fins were equally effective in cooling and were 80 percent better than the original steel fins.

INTRODUCTION

The problem of obtaining sufficient barrel cooling on modern high-output air-cooled engines has been the object of considerable investigation. Integral steel barrel fins have apparently approached a limit of maximum cooling imposed by manufacturing difficulties. Previous studies (reference 1) indicate that, for a given fin weight, considerably more cooling may be obtained with copper or aluminum fins than with steel fins. The chief obstacle to the adoption of copper or aluminum fins has been the difficulty of developing a completely satisfactory method of attaching the fins to the steel cylinder barrels.

The most successful production method so far developed for attaching fins has been the aluminum muff, which is an aluminum cylinder with integral fins, machined from a single aluminum billet. The muff is shrunk on the cylinder barrel. Objections to the use of the muff are the cost of manufacture and the limitations of fin thickness and spacing that can be machined on the muff.

Cylinder barrels having copper-plated steel fins have been tried on an air-cooled diesel-engine cylinder. A plating 0.008 inch thick on steel fins 0.043 inch thick resulted in about 11-percent improvement in over-all heat transfer. Copper plating, however, is limited in its range of possible improvement by the relatively poor fin proportions imposed by limitations in machining of the steel fins. For a given cooling effectiveness the copper-plated fins will weigh more than solid copper or aluminum fins.

Copper fins have been tested on barrels having electrical heaters inside the bore. These tests showed copper fins to be approximately as effective as aluminum fins on a weight basis and both were considerably better than steel fins. There are, however, several advantages that copper fins have when compared with aluminum fins. The higher thermal conductivity of copper permits the use of narrower fins than can be used with aluminum fins for a required degree of cooling, thereby interfering less with the air flow to the second bank of cylinders in a two-row or inline engine. Copper also has the advantage of being easily bonded to steel by means of a brazing alloy, thereby offering the possibility of a perfect bond. The type of finning described in this report has the additional advantage of not requiring a special thread belt on the barrel for attaching the cylinder head.

The object of this report is to show by power tests the improvement that can be obtained in cooling by the replacement of steel fins by copper fins. Comparative cooling tests were run on a standard Wright CGGC (G-200) cylinder on which the barrel fins had been replaced by a 1-inch copper fin wound spirally eight turns per inch. Improvements in cooling obtained in these tests were compared with the improvements predicted by the use of calculations shown in reference 2. An analysis was also made by computation to compare the cooling obtainable with the original steel fins and with copper and aluminum fins having the same weights as the original steel fins.

A study was also made to determine the effect of barrel cooling on piston temperatures. Obviously, as the barrel cooling is increased, the temperature drop through the wall increases and outside barrel temperatures are no longer a satisfactory index of engine cooling. Thermocouples were installed in the piston, and tests were run to determine the effect of barrel cooling on piston temperatures.

The tests were conducted at Langley Memorial Aeronautical Laboratory during April and May 1942.

CYLINDER CONSTRUCTION

The copper-finned cylinder used in these tests consisted of a copper-finned barrel screwed and shrunk into a stock cylinder head. The barrel was a stock Wright nitrided forging that was machined to accommodate the copper fins. The copper finning was applied by Buensod-Stacey Air Conditioning, Inc., in collaboration with the Induction Heating Corporation and Hardy & Harman.

The copper fins were applied in one continuous strip that had been preformed to the proper shape. An adequate length of strip was wrapped around the cylinder and pulled tight. The assembly was fluxed and a silver solder wire of the proper size and composition was laid between the fins. The heating of the assembly was accomplished by passing an induction heating head through the barrel at the proper speed. The speed of the operation was just sufficient to braze the spiral strip progressively to the barrel. The entire brazing operation required approximately 5 minutes and at no time was the nitrided barrel held at the brazing temperature long enough to damage the nitrided case.

After the brazing operation, the fins were work-hardened by a special tool. The finished barrel was sand-blasted to remove the excess flux.

The completed barrel was assembled with the head in the usual manner and the necessary machining operation required to complete the cylinder was performed.

APPARATUS AND METHODS

A standard Wright 094C (G-200) cylinder with steel fins shown in figure 1(b) was first set up on a single-cylinder test unit and complete cooling tests were conducted. The copper-finned cylinder (fig. 1(a)) was next set up and comparative tests were run. A section of the copper-finned cylinder showing the steel liner, the brazed bond, and the fin is shown in figure 2. A comparison of the dimensions of the original steel fins and the brazed copper fin is given in the following table:

	Width (in.)	Thickness (in.)	Fins/inch	Axial length of finning (in.)
Original steel	0.65	0.025	8½	5½
Copper	1.0	.037	8	5¼

In each case a stock Wright uniflow piston assembled with stock rings was used. The piston used with the copper-finned cylinder was equipped with thermocouples to measure temperatures at the crown, the ring belt, and at a point approximately midway in the skirt. The method of installation (reference 3) and the location of the piston thermocouples are shown in figures 3 and 4. The locations of the cylinder-surface thermocouples are shown in figure 5. The crankshaft used had a stroke of 7 inches, giving a displacement of 206 cubic inches.

The principal components of the test apparatus are shown diagrammatically in figure 6. The standard test-engine equipment was used for measuring brake mean effective pressure, engine speed, fuel consumption, cylinder temperatures, and fuel-air ratio. A separately driven centrifugal blower supplied the cooling air. The quantity of cooling air was measured by an 8-inch orifice at the entrance of a 12-inch-diameter pipe. The pressure tap for this orifice was 0.4 times the pipe diameter downstream, as recommended in reference 4.

Tests of both the copper-finned cylinder and the standard cylinder were conducted to determine the constants used in equations for the average barrel and head temperatures as functions of the fundamental engine and cooling variables by the methods described in reference 5. Tests were conducted in which each of the following was varied in turn while the remaining factors were held constant:

(1) Brake mean effective pressure

(2) Mass flow of cooling air

(3) Fuel-air ratio, F/A

The tests covered the following range of conditions:

Brake mean effective pressure, pounds per square inch	45-258
Cooling-air pressure drop across cylinder, inches of water	1-20
Fuel-air ratio	0.055-0.105

The friction horsepower was determined by motoring the engine at the test values of inlet-air pressure while the oil pressure, the oil temperature, the speed, and the cylinder temperatures were held at values as close as possible to those obtained under power conditions.

The fuel used in the tests was Army-Navy Fuel Specification No. AN-VV-NF-781 having an octane number of 100. For the more severe conditions, a sufficient amount of lead to prevent audible knock was added.

METHOD OF COMPILING DATA

Although the cooling tests were run with conditions as nearly the same on both cylinders as possible, it was impossible to maintain constant cooling-air temperatures. The cooling-air temperature varied from 79° F to 110° F, making it necessary to apply some correction to the test results before making a comparison. The test data were therefore compiled in the manner outlined in reference 5. The cooling characteristics of the cylinder barrel were expressed in terms of the equation

$$\frac{T_b - T_a}{T_g - T_b} = K \frac{I^n}{(\Delta P \rho / \rho_o)^m} \quad (1)$$

where

- T_b average outside barrel temperature, $^{\circ}\text{F}$
 T_a cooling-air temperature, $^{\circ}\text{F}$
 T_g effective temperature of gases in cylinder, $^{\circ}\text{F}$
 K proportionality constant determined by test
 I indicated horsepower
 ΔP cooling-air pressure drop across cowling, inches water
 ρ density of cooling air, taken as average of densities of air approaching and leaving fins
 ρ_0 standard air density, taken as density at 29.92 inches of mercury and 70°F
 n experimentally determined exponent
 m experimentally determined exponent
 Other symbols used in this analysis are
 H heat-transfer rate, Btu/hour
 T_h average head temperature, $^{\circ}\text{F}$, taken as average reading of head thermocouples indicated in figure 5

If T_g , K , n , and m are known, it is possible to compute barrel temperatures for a wide variety of operating conditions. From reference 5 it was found that, with a spark advance of 23°B.T.O. , and inlet-air temperature of 150°F , and a fuel-air ratio of 0.080, the values of T_g should be

Head	1200°F
Barrel	630°F

Variations of T_g with fuel-air ratio were computed from tests, as explained in reference 5. Constants for the basic heat-transfer equations were determined. From these equations the constants for the temperature correlation formula (1) were determined. Similar equations were developed for the rear spark plug and the exhaust valve seat. The equations thus obtained were used to compute the cylinder temperatures shown in later figures.

The method of correlating piston temperatures is explained in the discussion of results.

RESULTS AND DISCUSSION

Cylinder-Temperature Correlation

Figure 7 shows an example of uncorrected test data from the cylinders with the original steel fins and with the copper fins. The barrel with the copper fins was 60°F cooler at a power of 100 indicated horsepower. The cooling-air temperature, however, was 24°F hotter with the copper-finned cylinder. As a rough estimate, then, the reduction in barrel temperature resulting from the use of copper fins should be 84°F.

The various factors influencing cylinder temperatures were correlated in the form of curves shown in figures 8 through 12.

Figure 8 shows a comparison of the over-all coefficients of heat transmission for the original steel fins and the copper fins. The improvement resulting from the use of copper fins was 115 percent at 4 inches of water pressure drop. The theoretical increase in heat transfer as predicted by equations from reference 2 was 96 percent, a condition that shows that the improvements in cooling may be computed with reasonable accuracy.

The constants for the various cooling equations are obtained from the data shown in figures 8, 9, 10, 11, and 12. The general equation and constants are given below:

$$\frac{T_x - T_a}{T_g - T_x} = K \frac{(I/v)^n}{(\Delta P p / \rho_o)^m}$$

where

T_x temperature at a given point on cylinder, °F

I/v specific power output, indicated horsepower per cubic inch of displacement

ΔP cooling-air pressure drop across baffle, inches of water

The value of T_g is obtained from figure 9 and the values of K , m , and n are listed in the following table:

<u>Location</u>	<u>K</u>	<u>n</u>	<u>m</u>
Barrel, copper fins	0.770	0.59	0.31
Barrel, steel fins	1.560	.59	.27
Rear spark plug	1.085	.65	.27
Exhaust valve seat	1.110	.50	.22

Piston-Temperature Correlation

Figure 11 shows the results of tests to determine the effect of barrel temperature on piston temperature. Each curve represents data taken at constant power output. The barrel temperature was varied by changing the cooling-air pressure drop. It is shown that a slope of -0.5 may be used to represent the relation between $T_p - T_b$ and T_p for the range of temperatures involved. The relation between piston temperature and barrel temperature was therefore assumed to be

$$T_p - T_b = K + nT_p \quad (2)$$

where n is slope of the curve at constant power representing the relation between $T_p - T_b$ and T_p

T_p piston-ring-belt temperature, $^{\circ}\text{F}$

T_b average barrel temperature, $^{\circ}\text{F}$

K an experimental function of power, inlet-air temperature, fuel-air ratio, spark advance, and compression ratio

$$\text{Therefore } T_p - T_b = K - 0.5 T_p \quad (3)$$

For convenience the constant K is divided into two factors

$$T_p - T_b = fG - 0.5 T_p \quad (4)$$

Factor G is the constant K when the fuel-air ratio is 0.080, the inlet-air temperature is 150° F, and the spark timing is 22° B.T.C. Variations in cooling resulting from variations in fuel-air ratio, inlet-air temperature, and spark timing are represented by f . Figure 12(b) shows the variation of fG with power computed from piston temperatures taken simultaneously with the barrel-cooling tests. The effect of fuel-air ratio is immediately apparent. From these curves the following values of f are obtained:

<u>F/A</u>	<u>f</u>
0.060	1.034
.080	1.000
.100	.910

The failure of a piston thermocouple at the termination of the barrel-cooling tests necessitated a disassembly of the engine. When the engine was reassembled the values of fG were slightly altered to about 30° F less at 150 indicated horsepower, representing a decrease in piston temperature of about 20° F at that power.

The relation between G and power can be represented by the equation

$$G = AI^r$$

where

A proportionality factor

r experimentally determined exponent, equal to slope of curves in figure 12(b)

G correlation factor fG when $F/A = 0.080$

For the data of the barrel-cooling tests the constants were found to give the following equations

$$G = 55.1 (I^{0.44}) \quad (6)$$

$$G = 574 (I/\tau)^{0.44} \quad (7)$$

Substituting equation (7) in equation (4)

$$T_p - T_b = 574 (I/v)^{0.44} - 0.5 T_p$$

$$T_p = \frac{T_b + 574 (I/v)^{0.44}}{1.5} \quad (8)$$

Comparative Performance

The rated power of an engine using C9G0 cylinders is approximately 0.7 indicated horsepower per cubic inch. Figure 13 shows that at this power output and a cooling-air pressure drop of 16 inches of water the average temperature is 213° F on the copper-finned barrel, as compared to 297° F on the steel-finned barrel. The corresponding piston-ring-belt temperatures are 467° F and 525° F. The piston temperature in this case changes 0.70° per degree of change in average barrel temperature.

Figure 13 also shows the decrease in required cooling-air pressure when the original steel fins are replaced by copper fins. For example, if it is desired to maintain a piston-ring-belt temperature of 525° F, the steel-finned barrel will require 16 inches of water, as compared with 1.2 inches of water for the copper-finned barrel.

Figure 14 extends the comparison of cylinder temperatures to higher and lower power outputs when the cooling-air pressure is 16 inches of water.

A comparison of the changes in piston and barrel temperatures as the result of improved fin design shows that the outside barrel temperature may be a poor criterion for barrel cooling. For example, at 0.7 indicated horsepower per cubic inch, the average temperature of the steel-finned barrel is 297° F, and of the piston, 525° F. If the power of the copper-finned cylinder were raised until the barrel temperature reached 297° F, the piston temperature would be well above 600° F and scuffing of the rings would be probable. It is therefore evident that the maximum outside barrel temperature should be reduced if the power is increased. Figure 15 shows the influence of power on the barrel temperature required to maintain a constant piston-ring-belt temperature of 525° F.

At the time of the preparation of this report, the copper-finned cylinder had completed 60 hours of operation with the brake mean effective pressure varying from 45 to 258 pounds per-square-inch. For the greater percentage of this time the engine was operated at an average brake mean effective pressure of 200 pounds per square inch. A close inspection of the finning did not reveal any structural weakness. The condition of the piston rings and the cylinder bore was particularly good.

Copper and Aluminum Fins Having the Same Weights as the Original Steel Fins

The proportions of the copper fins used in these tests were selected solely for ease of fabrication and assembly. The best fin proportions will be considerably different from those selected for this test. These dimensions for a given cylinder are governed by many considerations that cannot be properly evaluated in this study. In some cases it is desired to use copper or aluminum fins having the same weight as the steel fins now used. A theoretical study is therefore given here to show the improvements in cooling that can be obtained from copper or aluminum fins having the same weights as the original steel fins on the Wright CGO cylinder.

Figures 16 and 17 show the improvements in computed coefficients of over-all heat transfer, U , Btu/(sq in.) ($^{\circ}\text{F}$)(hr), from the outside of the barrel to the air, when copper fins having the same weight as the original steel fins are used. The greatest improvement is 2.17 times the heat transfer from the original steel fins and is obtained with fins 1.4 inches wide and 0.0056 inch thick. Such thin fins might be considered to be impractical because they would have insufficient mechanical strength and because much thicker fins may be used with little sacrifice in cooling. At present the practical minimum fin thickness may be considered to be in the range from 0.010 inch to 0.020 inch thick. In this range the improvements in heat transfer with proper fin width and spacing are 1.8 to 2.1 times the heat transfer from the original fins and with no increase in fin weight.

Figure 18 shows the improvements obtainable from aluminum-muff fins having the same weights as the original steel fins. The range of practical fin dimensions in the

aluminum-muff design is restricted by limits imposed in machining the fins. With aluminum fins, as with copper fins, the fin thickness and spacing may vary considerably from optimum without serious loss in heat transfer.

Representative designs of copper and aluminum fins are compared below. In each case the dimensions were considered to be as near optimum as is practical to fabricate by modern methods of production:

Fins ---	Width (in.)	Thickness (in.)	Spacing (in.)	U/U_{steel}
Original steel	0.65	0.025	0.093	1.00
Copper	1.10	.018	.155	1.85
Aluminum	1.10	.025	.106	1.78

Roughly, the copper and aluminum fins give the same cooling, which is about 1.8 times the cooling of steel fins, provided that the weights of the fins are equal.

CONCLUSIONS

From the comparative cooling tests on a standard cylinder with steel fins on the barrel and on a cylinder with copper fins on the barrel, it may be concluded that:

1. The 1-inch copper fins had a 115-percent greater over-all coefficient of heat transfer than the original steel fins. This improvement resulted in a reduction of average barrel temperature at rated maximum continuous power from 297° F to 213° F.

2. The improvement to be obtained with better fins may be computed with reasonable accuracy by the use of the method shown in reference 2.

3. Computations show that, in the range of practical fin dimensions, copper fins having the same weight as the original steel fins will give at least 1.8 times the over-all heat transfer of the original steel fins.

4. In the range of practical fin dimensions, copper- and aluminum-muff fins having the same weights will give approximately the same over-all heat transfer.

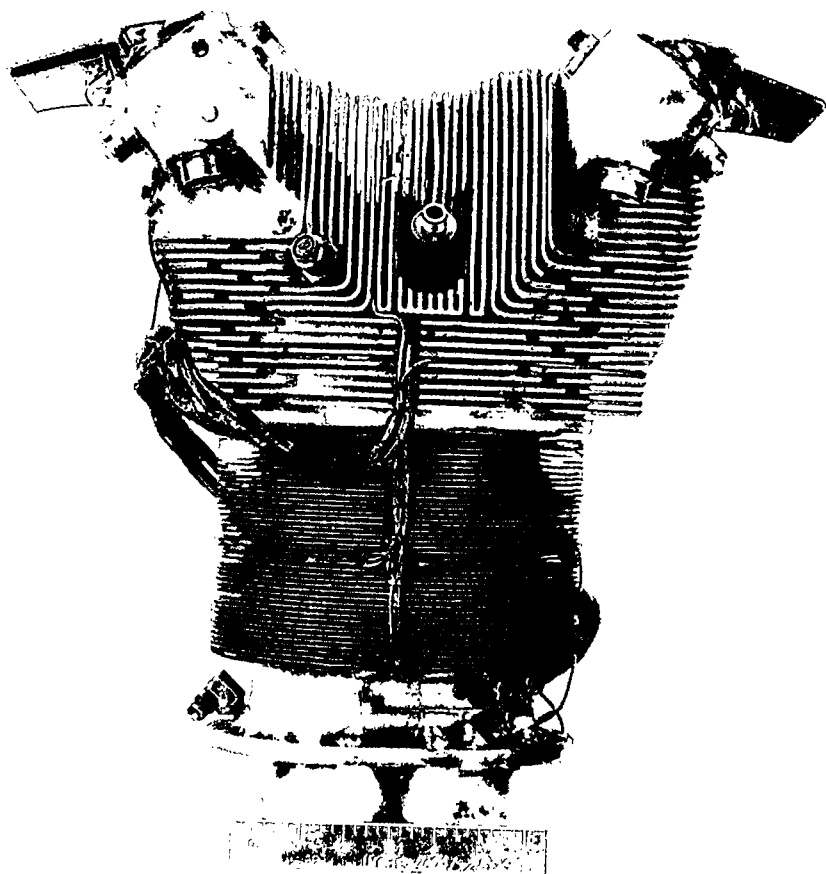
5. The piston-ring-belt temperature is believed to be a more accurate criterion than barrel temperatures for required engine cooling.

6. For equal piston-ring-belt temperatures, a lower maximum permissible barrel temperature will be required when improved cylinder barrel cooling is provided.

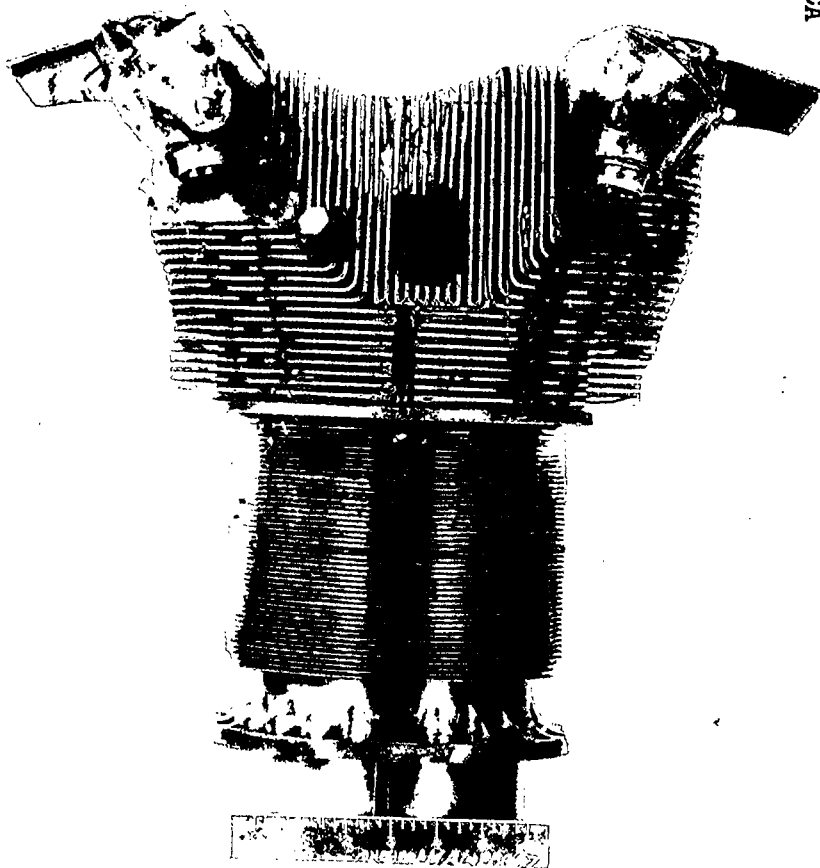
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Langley Field, Va.

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3. Pinkel, Benjamin, and Manganiello, Eugene J.: A Method of Measuring Piston Temperatures. TN NACA No. 765, 1940.
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5. Pinkel, Benjamin, and Ellerbrock, Herman H., Jr.: Correlation of Cooling Data from an Air-Cooled Cylinder and Several Multicylinder Engines. NACA Rep. No. 683, 1940.



(a) Copper-finned barrel.



(b) Standard C9GC barrel.

Figure 1.- Photograph of Wright C9GC cylinders with original steel fins and with 1-inch spiral copper fins.

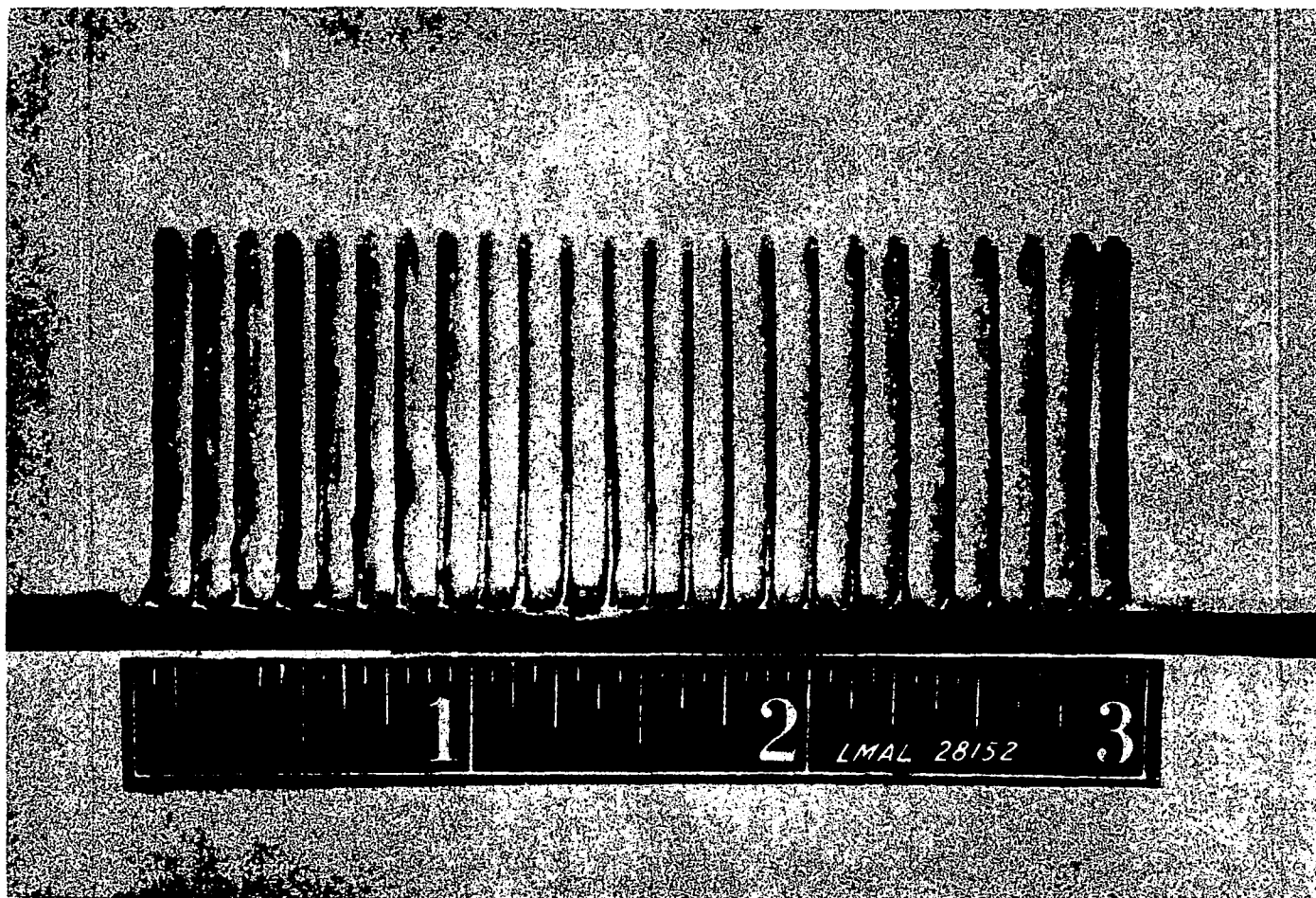
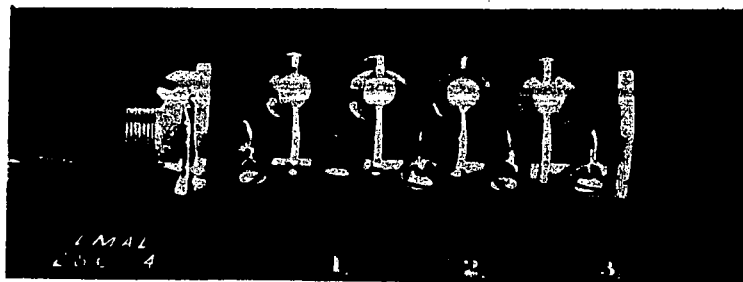
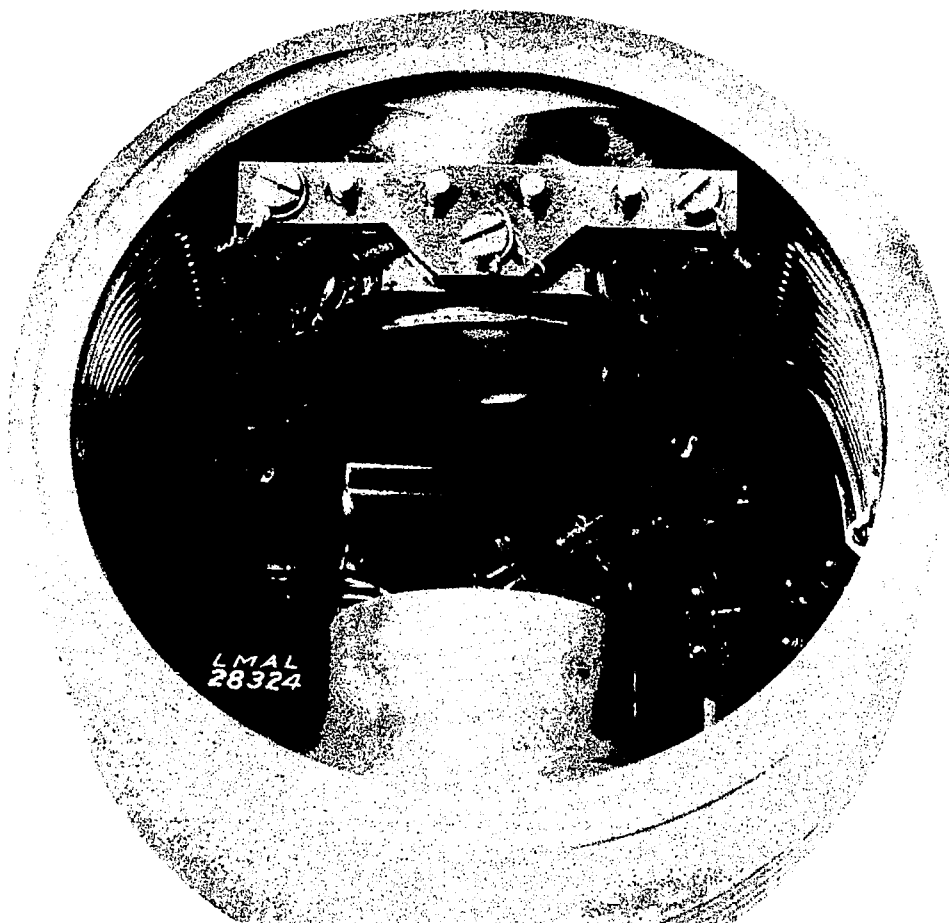


Figure 2.- Photograph of a cross section of the copper fins.



(a) Plunger block with contacts.



(b) Piston with thermocouple installation and contacts.

Figure 3. - Piston-thermocouple installation.

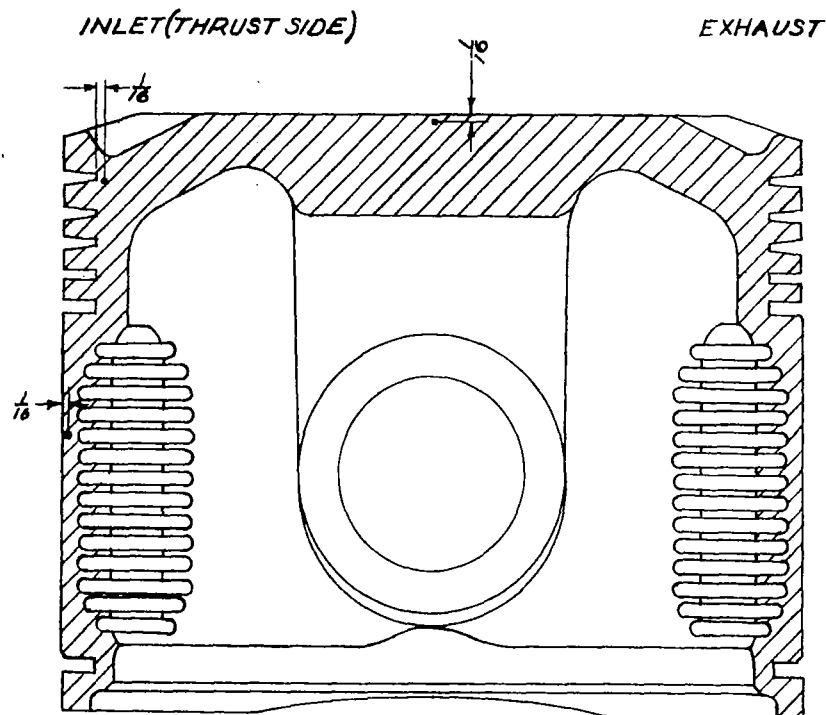


FIGURE 4. - LOCATION OF PISTON THERMOCOUPLES.

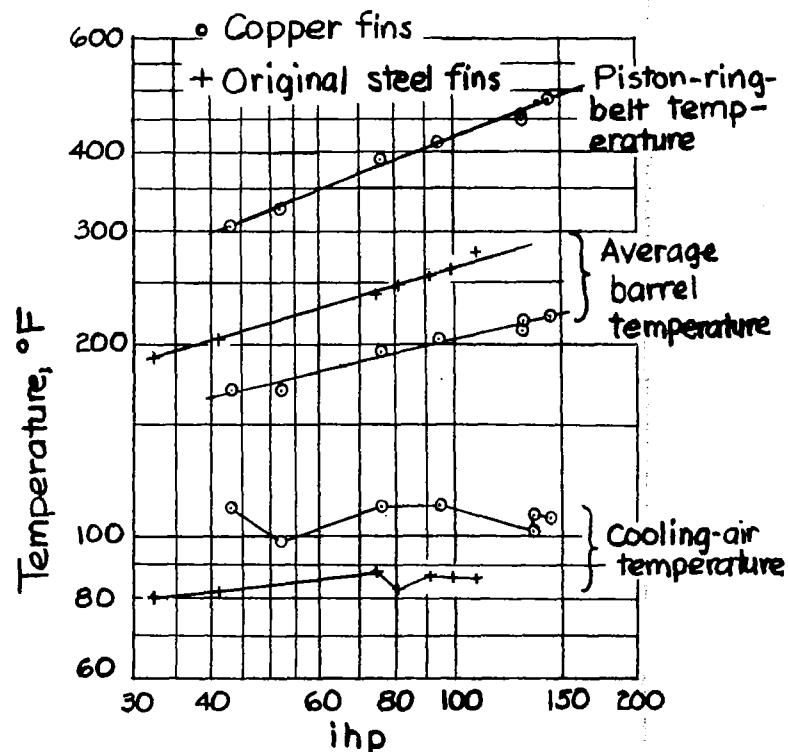


Figure 7. - Examples of uncorrected test data. Inlet-air temperature, 150°F; engine speed, 2000 rpm; cooling-air pressure drop, 16 inches of water; fuel-air ratio, 0.080.

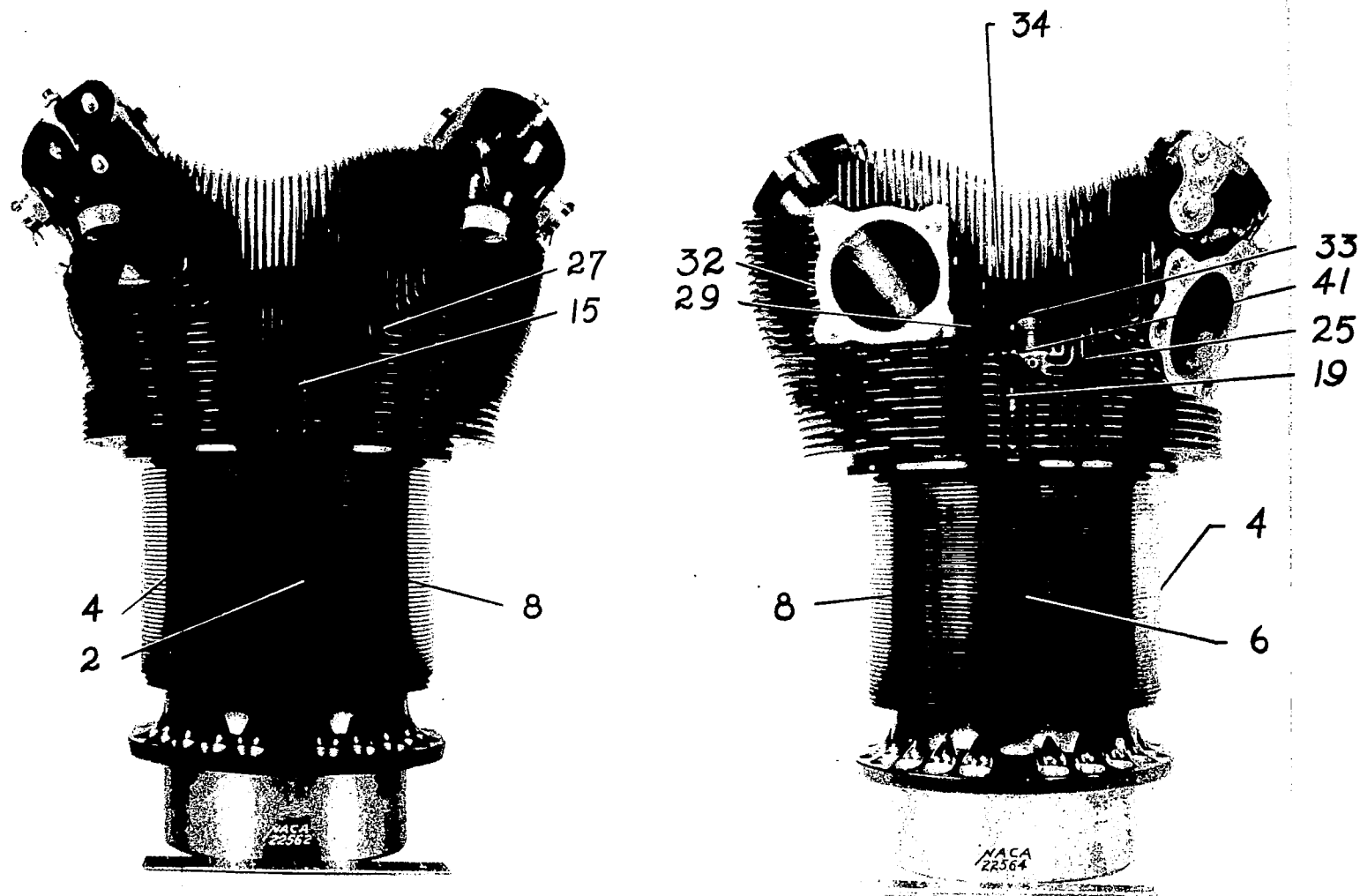


Figure 5. - Location of surface thermocouples.

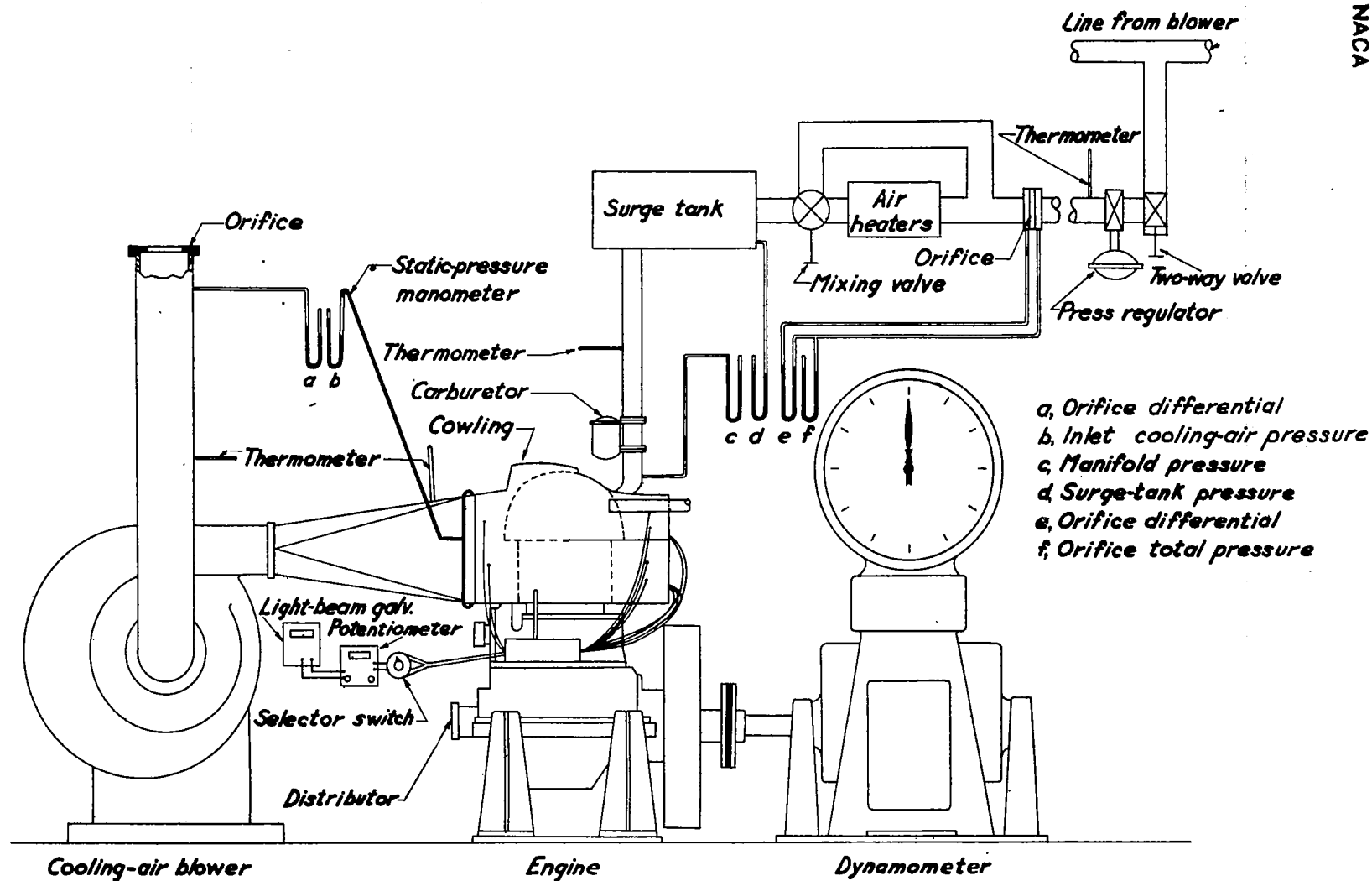


Figure 6.-Diagrammatic layout of equipment.

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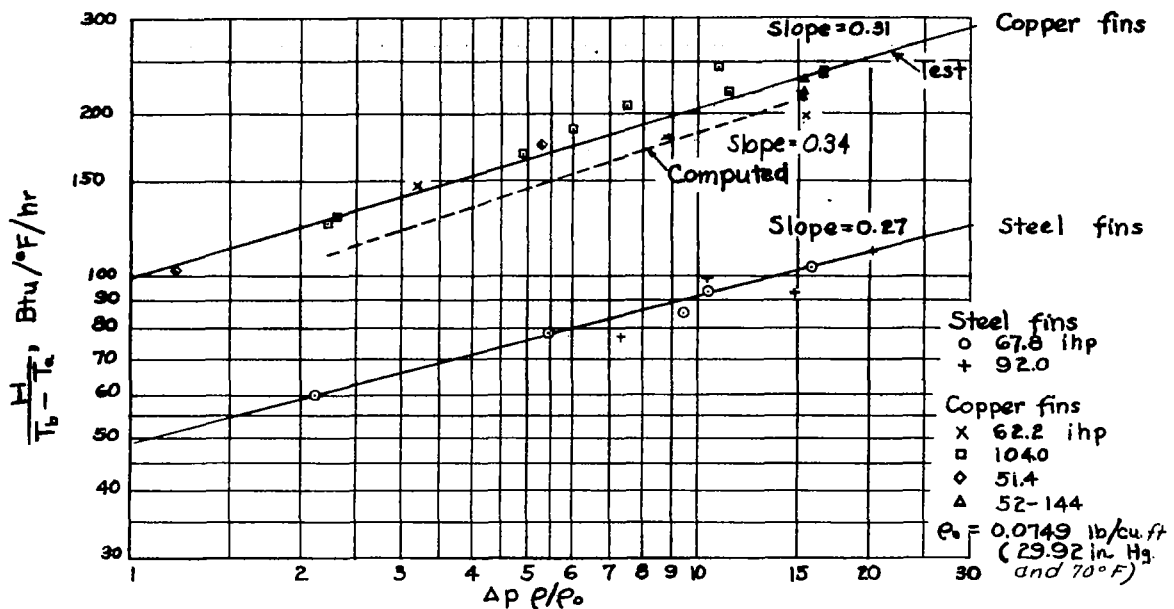


Figure 8.-Comparison of heat-transfer characteristics of original steel fins and copper fins. Wright C9GC cylinder.

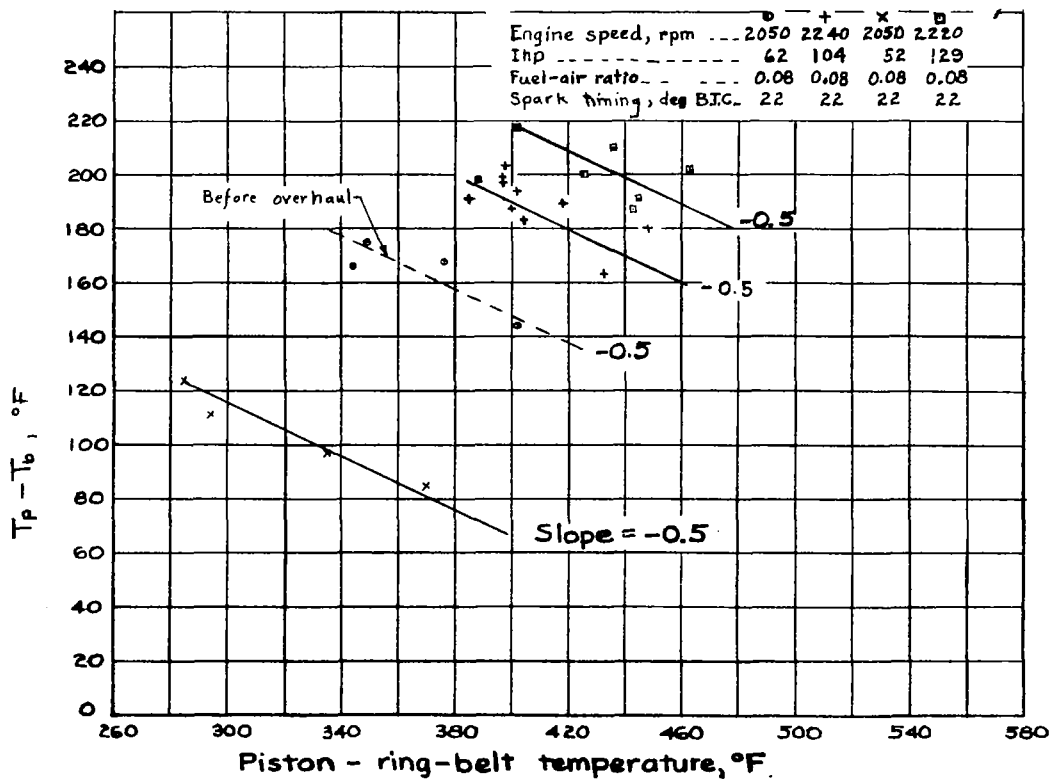


Figure 11 - Influence of barrel temperature on piston temperature.

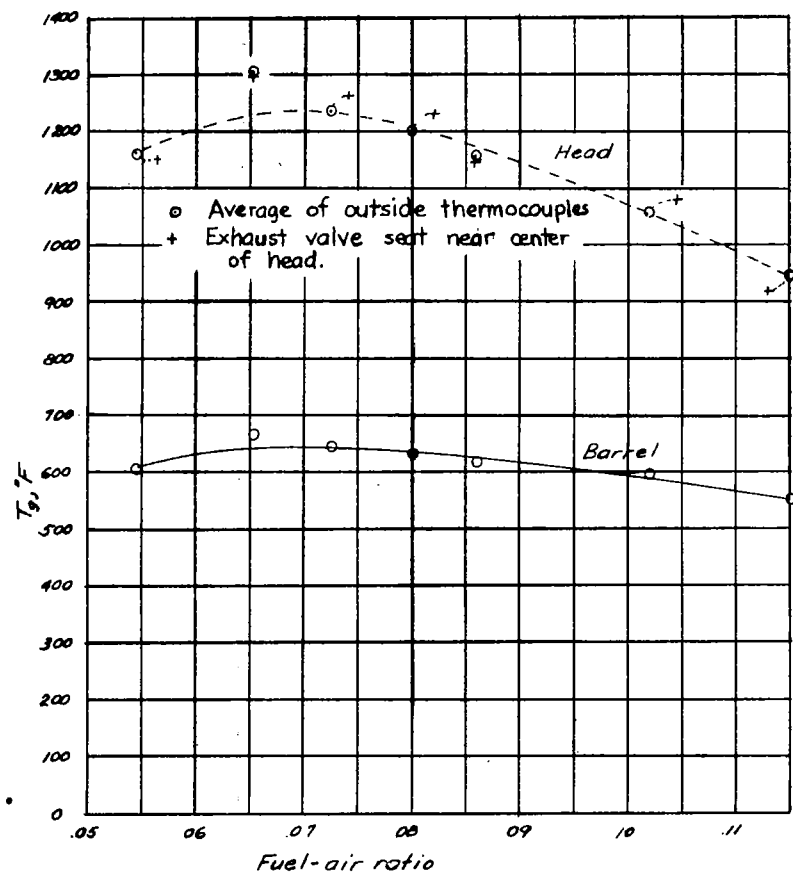
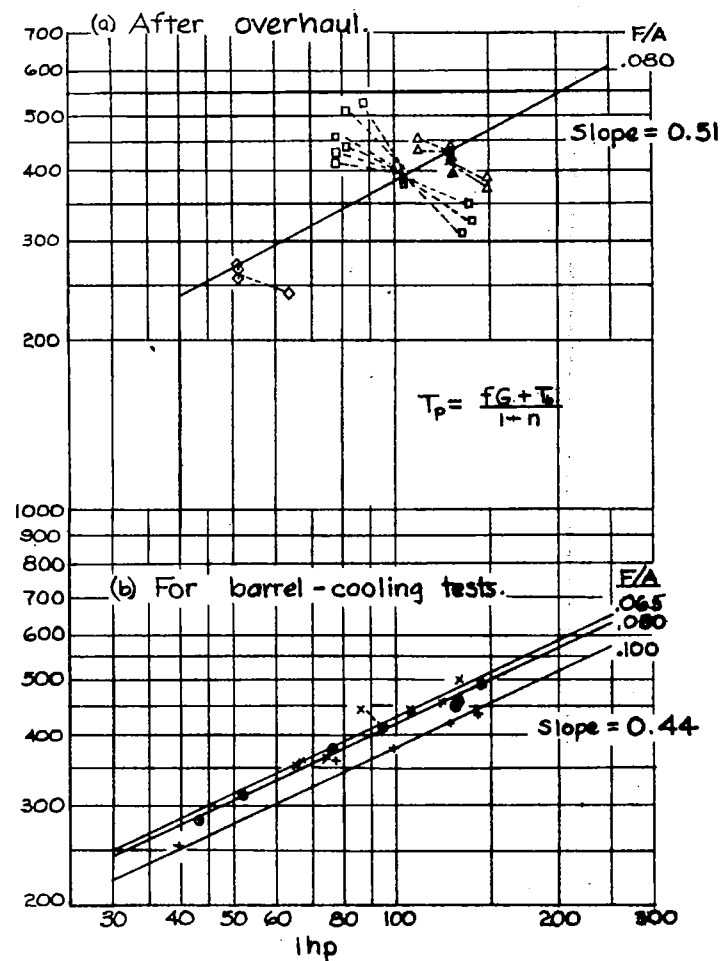
Figure 9. - Effect of mixture ratio on T_g Correlation temperature, fG , °FCorrelation temperature, fG , °F

Figure 12. - Piston - temperature correlation. C9GC cylinder.

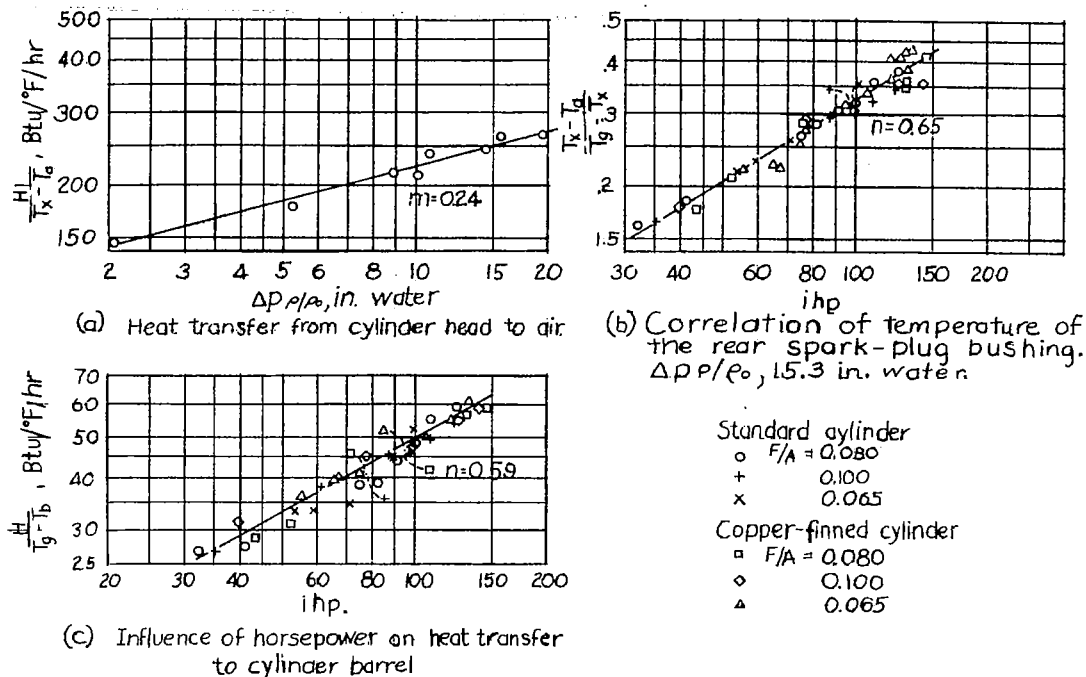


Figure 10.- Correlation of cylinder temperatures.

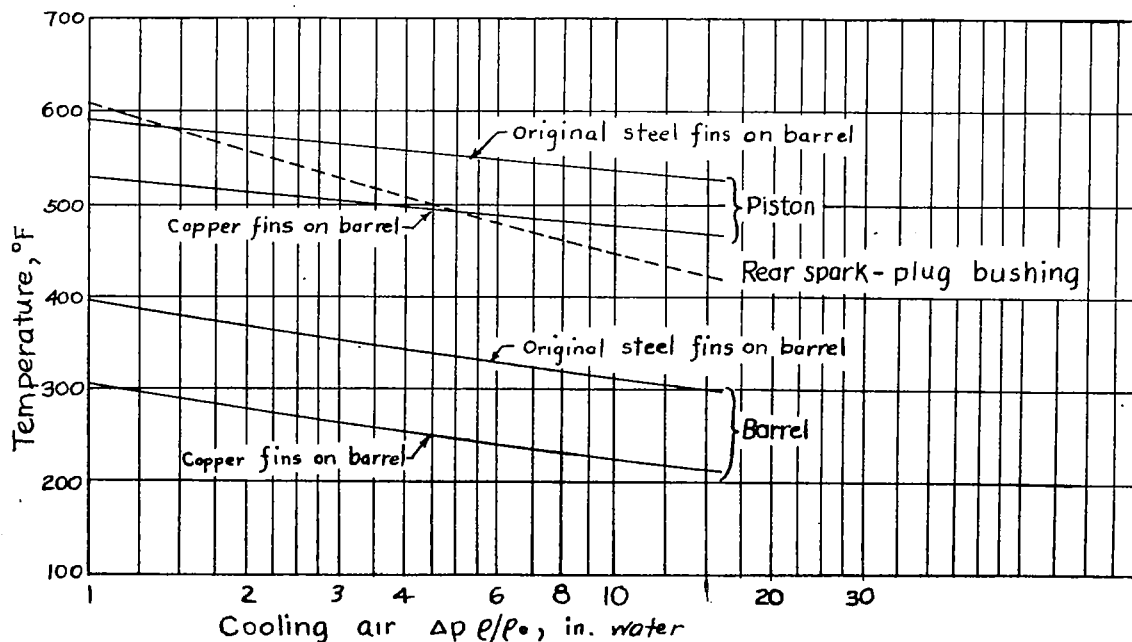


Figure 13.- Effect of cooling-air pressure drop on cylinder temperatures. Indicated horsepower per cubic inch, 0.070; cooling-air temperature, 100°F; inlet-air temperature, 100°F; fuel-air ratio, 0.080.

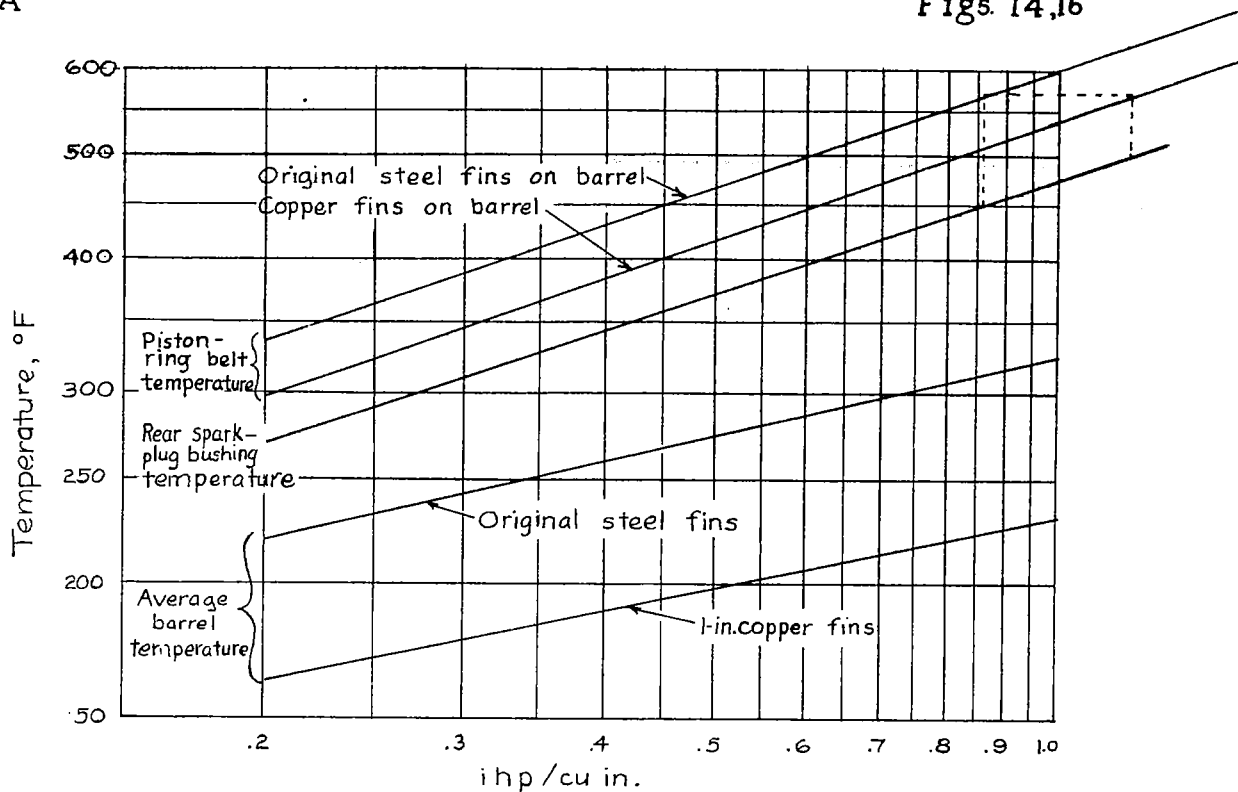


Figure 14.- Effect of improved barrel cooling on cylinder temperatures.

Cooling - air temperature, 100°F; inlet-air temperature, 150°F;
 Fuel-air ratio, 0.08; Δp ρ/ρ_0 , 16.0 in. water.

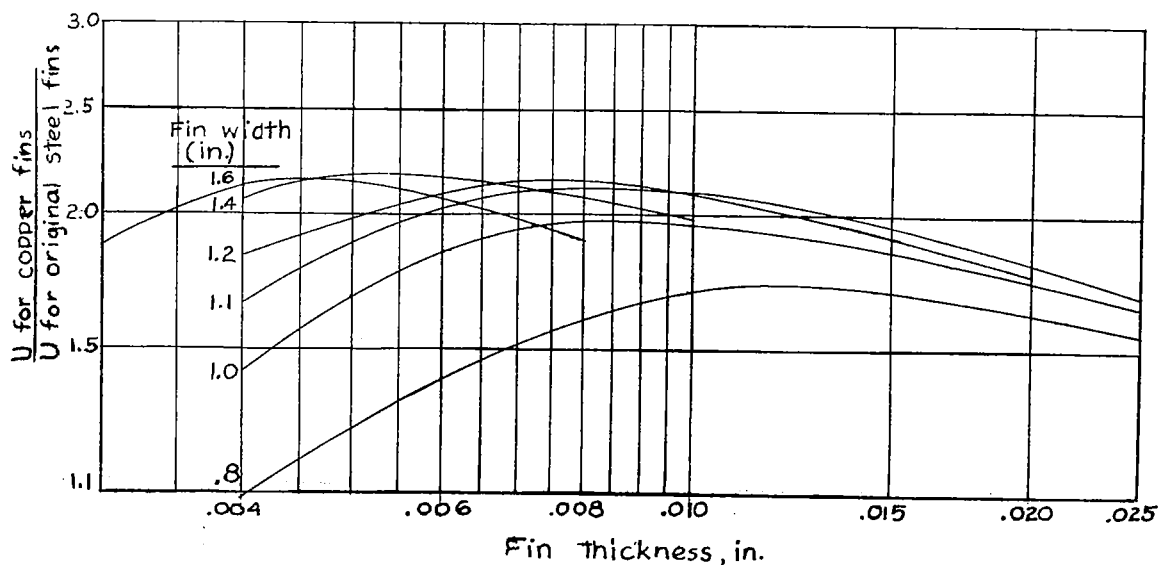


Figure 16.- Influence of fin dimensions on the heat-transfer coefficient of copper fins having the same weight as the original steel fins on a C96C cylinder barrel.

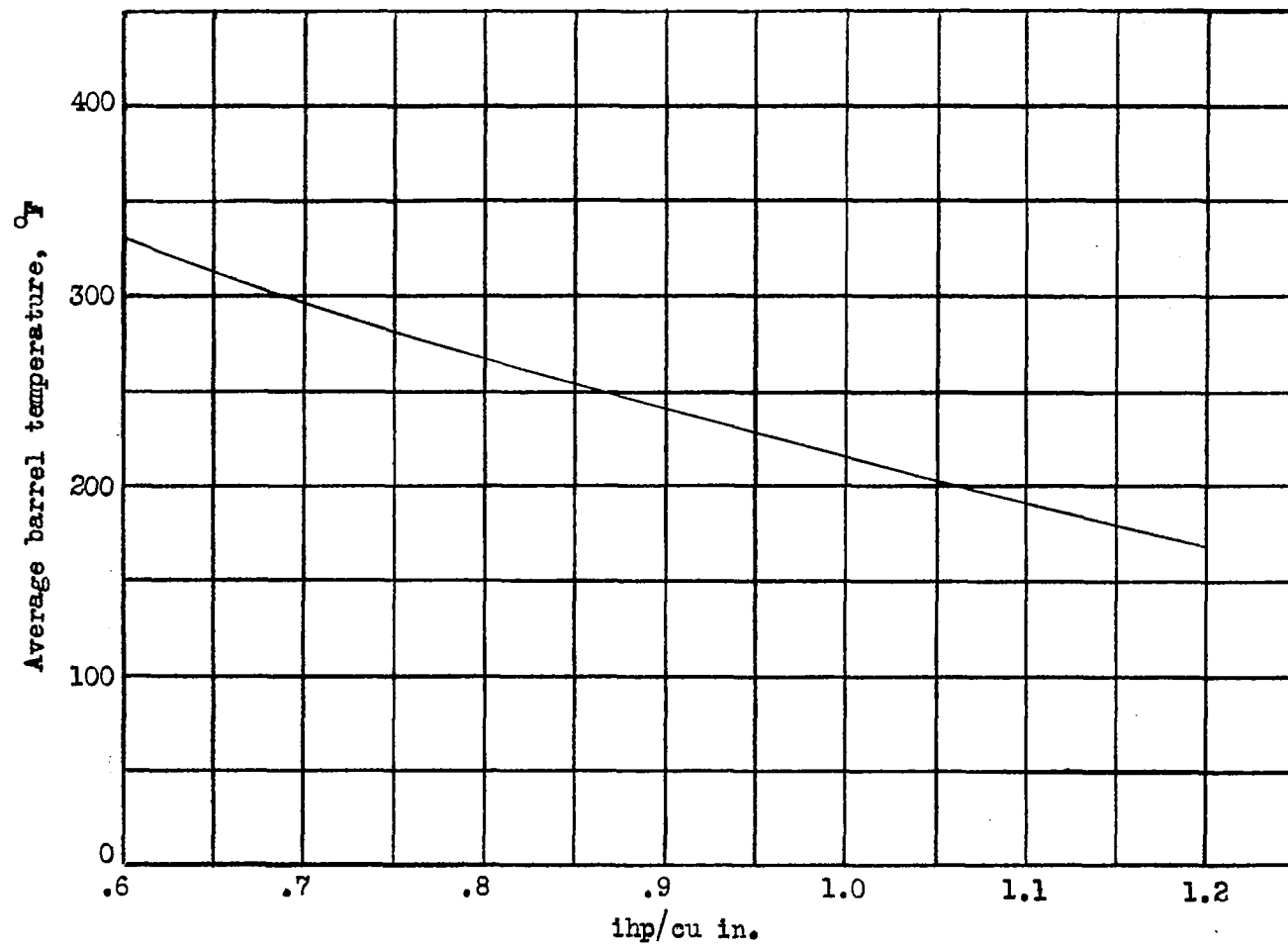


Figure 15.- Average barrel temperature required to maintain a piston-ring-belt temperature of 525° F. Inlet-air temperature, 150° F; fuel-air ratio, 0.080.

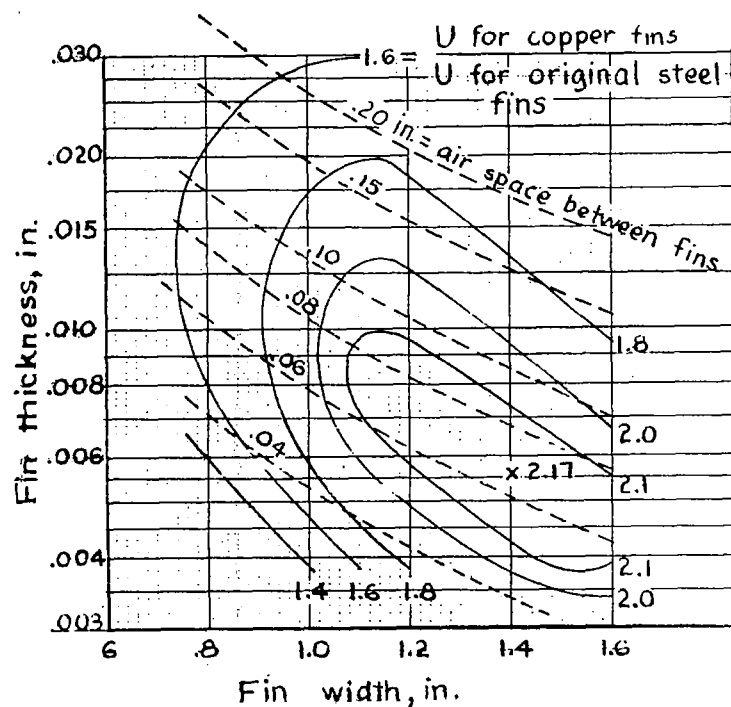


Figure 17.- Sizes of copper fins having the same weight as the original steel fins on a Wright C96C cylinder barrel.

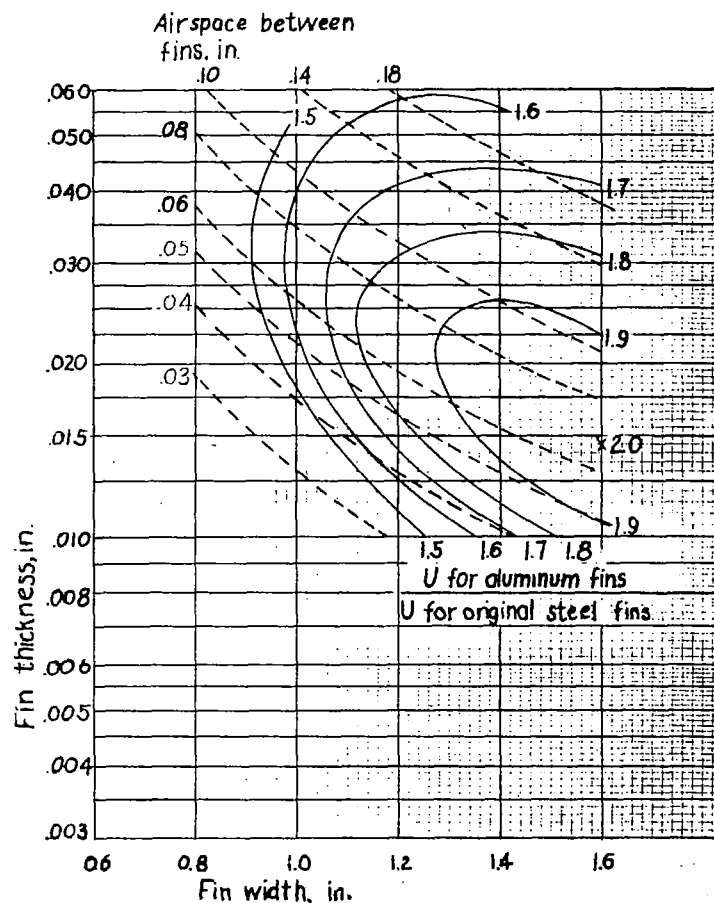


Figure 18.- Sizes of aluminum-muff fins having the same weight as the original steel fins on a Wright C96C cylinder barrel. Sleeve of muff, 3/32 inches thick; weight includes sleeve.

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